HEAD LOSS In Quick-Coupled Aluminum Pipe

Used For Sprinkler Irrigation Systems

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HEAD LOSS in Quick-Coupled Aluminum Pipe

Used for Sprinkler Irrigation Systems

By W. O. Ree, *project supervisor, Soil and Water Conservation Research Division, Agricultural Research Service*

This handbook presents a study of current practices in estimating friction loss in portable aluminum-pipe sprinkler irrigation systems. Literature by manufacturers of sprinkler irrigation equipment has been reviewed by the author. Reports of experiments to determine losses in irrigation pipe and fittings have been studied, as well as theses on the subject. Conclusions reached by analyzing this material are presented for the information and guidance of those engaged in designing and selecting irrigation systems.

Field checks have disclosed that actual performance of sprinkler systems has not always measured up to that which was predicted. Indications are that head loss, as a rule, has been greater than calculated. When such differences have been found, head loss resulting from pipe friction has been regarded as the most probable cause. Factors used in estimating head loss have been based on the condition of the pipe when new. It has been determined, however, that under some circumstances aluminum pipe corrodes while in use; also, that frequent moving dents the pipe. This corroding and denting increase the roughness of the pipe and thereby increase friction loss. A point to be determined in the investigation has been whether enough increased friction has resulted to account for the difference between actual and estimated head loss. If so, more realistic values of the friction factor should be used in designing irrigation systems. Existing experimental evidence has been examined, in an effort to find the solutions to these problems.

Another source of head loss involved in the problem studied is that caused by couplers. Recently, data have been obtained on loss coefficients for the quick couplers used in aluminum-pipe systems. These coefficients have a considerable range, depending on the design of the coupler and the deflection angle between the pipes joined at the coupler. Some of the coefficient values included in these data are quite high.

This has raised a question as to whether pipe-friction-loss tables in current use include coupler loss; and if so, whether the values used are reasonable. Manufacturers' literature and available bulletins and handbooks have not been particularly helpful in settling these questions. In fact, various sources have been contradictory. Basic experimental data have therefore been used in arriving at conclusions.

This study has disclosed that three formulas expressing the relationship between rate of flow, pipe size, and head loss were already in general use. These were the Scobey formula, the Manning formula, and the Hazen-Williams formula. Actually, none of these was employed in the study of the experimental data. Instead, the Darcy-Weisbach formula was used. Relationships were then worked out to convert the Darcy-Weisbach f values to coefficients for the more common formulas. These relationships also provide a means for determining equivalent values for the different types of coefficients. Diagrams showing these relationships are included in this handbook.

DEFINITIONS OF SYMBOLS

Definitions of symbols used in this handbook are as follows:

- a A constant, a function of boundary condition—determined by the form of the coupler.
- C_1 Coefficient in the Hazen-Williams formula.
- D Diameter of pipe (feet).
- d Diameter of pipe (inches).
- F The Christiansen F factor for estimating head loss in a line with multiple outlets.
- f Darcy-Weisbach friction factor.
- g Acceleration of gravity (32.2 feet per second).
- H_f Head loss due to pipe friction (feet).
- h_c Head loss in the pipe coupler (feet).
- K A constant in a general head-loss equation.
- K_c Head-loss coefficient for pipe coupler.
- K_s Coefficient in the Scobey formula.
- L Length of pipe over which head loss is determined (feet).

m An exponent in a general head-loss equation.
n The Manning roughness coefficient.
n An exponent in a general head-loss equation.
Q Discharge rate (cubic feet per second).
R Ratio of diverted flow to the total flow approaching a side outlet.
R Hydraulic radius (feet).
R_e Reynolds number.
S Slope of the pressure grade line.
V Flow velocity (feet per second).
V_1 Velocity of flow approaching a side outlet (feet per second).
Δz Pressure rise at a side outlet (feet).
ϵ The absolute roughness height in the Colebrook-White equation. (The unit is the same as for diameter, *D*.)

PRESENT DESIGN STANDARDS

A review of manufacturers' literature has revealed that the basic reference usually presented for the calculation of head-loss tables was a bulletin by Christiansen (*1*).[1] That part of this bulletin which deals with pipe-friction factor will therefore be discussed first. This bulletin, which was written in 1942, does not mention aluminum pipe. However, friction loss in welded-steel pipe of the type commonly used for main supply lines for portable sprinkler systems is discussed. A graph of friction loss in welded outside diameter (O. D.) pipe is given (*1, p. 61*). Accompanying this diagram is the explanation: "This graph is based on Scobey's formula, with the coefficient K_s=0.32, which corresponds to new pipe in good condition." In a discussion of friction loss in sprinkler lines (*p. 67*), he states: "Most portable sprinkler pipe is made from lightweight O. D. tubing." Here also he apparently means steel pipe. A diagram (*p. 68*) shows friction loss in portable sprinkler pipe. This is based on the Scobey formula

$$H_f=\frac{K_s L V^{1.9}}{1{,}000\, D^{1.1}}$$

Where

H_f is the total friction loss in a length of pipe (feet)
L is the length of pipe (feet)
V is the mean velocity (feet per second)
D is the diameter of the pipe (feet)
K_s is the Scobey coefficient.

Values of K_s suggested by Christiansen are given in table 1.

TABLE 1.—*Values of* K_s,[1] *based on the Scobey formula*

Diameter of O. D. pipe	Wall thickness	K_s values
Inches	*Gage*	
1	18	0.38
1½	18	.36
2	18	.35
2½	18	.34
3	16	.33
4	16	.32
5	16	.32
6	16	.32

[1] Suggested by Christiansen (*1*).

[1] Italic figures in parentheses refer to Literature Cited, p. 20.

This statement is made by Christiansen (*1, p. 67*): "According to tests on such pipe, higher values of K_s may be expected of the smaller sizes, and the values are influenced by the type of coupling. The values given are believed conservative; most sprinkler pipe will have less friction loss than is shown by the graph." The foregoing statement implies that coupler loss is included in the values of K_s.

Christiansen also developed an approximate method for estimating friction loss in a pipe with multiple outlets. The method involves calculating a head loss for the entire flow through the entire length of pipe. A factor (*F*) is then applied to this figure, to arrive at the head loss for the actual flow condition. Factor *F* varies with the exponent (*m*) in the following equation:

$$H_f=F\frac{KLQ^m}{D^{2m+n}}$$

If m=2, factor *F* will vary from 1.0 for 1 outlet on the line to 0.359 for 20 outlets and 0.338 for 100 outlets. The approximation in this method is the assumption that all outlets have equal discharge rates. This is nearly true in a well-designed sprinkler system.

The textbook of the Sprinkler Irrigation Association (*6*) is an important work in this field, for it represents the industry generally. Pertinent statements on the subject of pipe friction in this textbook are quoted or paraphrased in the following paragraphs.

"Friction values now used throughout the industry have been checked fairly close under field conditions and are found to be on the conservative side." (See *6, p. 247.*) No corroborative material is given to support the statement.

Scobey's formula is presented with the note that it was developed for welded-steel and similar pipe. Christiansen's suggested values for K_s are listed. It is then stated that when aluminum pipe was introduced, designers continued to use Scobey's formula and Christiansen's K_s values. (See *p. 253.*)

The following statement is made on the subject of couplers: "To handle this problem [added coupler loss] * * *, value of K_s in Scobey's formula was increased for portable aluminum main line [to] K_s=0.40. Most charts and slide rules used by the industry are based on this value." (See *p. 255.*)

Reference is then made to the work of Gray and coworkers (*3*) at Cornell University: "Based on this preliminary data, it would appear that charts or tables based on Scobey's formula and $K_s=0.40$ are on the conservative side, at least for most commonly used coupler types. Again, until data is complete, the writers suggest using existing material on the basis of the Scobey formula. It should again be pointed out that aging and corrosion in some areas may make values, based on the Scobey formula, nearly correct for the average life of main lines." (See *p. 255.*)

Table X–5 of the cited textbook gives loss of head in feet per 100 feet of plain aluminum pipe. The table is based on the Scobey formula, with $K_s=0.34$ for 2-inch pipe, 0.33 for 3-inch pipe, and 0.32 for other sizes. It is noted that the values are for aluminum pipe *without* couplers. (See *p. 422.*)

EXPERIMENTS ON FRICTION FACTOR

Experiments on friction factor for aluminum tubing were located at several State universities. These limited data, in most instances, were incidental to the main objective of the experiment, which usually involved measuring fitting losses. Nevertheless, these data have proved useful and valuable in this study.

Each experimenter has portrayed his results in his own fashion. Most have chosen to present an equation relating head loss to velocity and diameter. The equation would fit the data for the particular experiment but would not be of great usefulness for comparison with other experiments. In one instance, the experimenter also calculated the Darcy-Weisbach friction factors for the individual runs. In two other experiments, no head-loss equations or friction-factor values were calculated, even though the data needed for doing so were available.

A uniform method of analyzing and comparing all the data had to be selected. The Darcy-Weisbach formula

$$H_f=f\frac{L}{D}\frac{V^2}{2g}$$

was chosen for this purpose. The value of the friction factor f and the corresponding Reynolds number for the flow were calculated for each run. The data were then plotted on log paper. A separate plotting was made for each experiment. It was noted that the data usually followed the pattern found in the familiar pipe-friction-resistance diagram. An example of this diagram can be found in King's Handbook (*4, sect. 6, p. 8*).

Experiments with fluid friction in pipes have disclosed that the data for full turbulence usually fit the Colebrook-White transition curves. These are given by the equation

$$\frac{1}{\sqrt{f}}=-2\log\left(\frac{\epsilon}{3.7D}+\frac{2.51}{R_e\sqrt{f}}\right).$$

When ϵ, the absolute roughness height, becomes zero, the equation reduces to the Karman-Prandtl expression for turbulent flow in smooth pipes. It is

$$\frac{1}{\sqrt{f}}=2\log R_e\sqrt{f}-0.8.$$

The foregoing equations are used as standards for comparison in this publication. The Karman-Prandtl equation is shown on each plotting. Where the pipe is not hydraulically smooth, Colebrook-White curves with appropriate $\frac{\epsilon}{D}$ ratios are added. These curves, when plotted with the data, provide a quick way of evaluating hydraulic properties of pipe and a means of comparing the various experiments.

Each experiment studied is described separately in subsequent sections.

Research by H. E. Gray, G. Levine, and M. Bogema

Gray and coworkers (*2*) present the results of friction-loss determinations on two 20-foot lengths of 3-inch aluminum pipe. The pipe had been used for 3 seasons in field irrigation. Two sets of 20 flow rates were used in the experiment with Reynolds number, ranging from 4.82×10^4 to 1.36×10^5. The results are given in the following 3 equations:

$$\text{Upstream pipe } H_f=0.30\frac{L}{1{,}000}\frac{V^{1.71}}{D^{1.29}}$$

$$\text{Downstream pipe } H_f=0.30\frac{L}{1{,}000}\frac{V^{1.77}}{D^{1.23}}$$

$$\text{Composite } H_f=0.30\frac{L}{1{,}000}\frac{V^{1.74}}{D^{1.26}}$$

Gray's formula can be converted to a form having the Darcy-Weisbach f and Reynolds number as the variables. For a water temperature of 60° F., the equation becomes

$$f=0.3668\,R_e^{-0.26}.$$

The original data for the experiment were obtained from Professor Gray, and the Darcy-Weisbach f values were calculated. These were plotted against the corresponding Reynolds numbers (fig. 1). A comparison of the plotted points with the calculated line derived from the Karman-Prandtl equation shows the pipe to be hydraulically smooth.

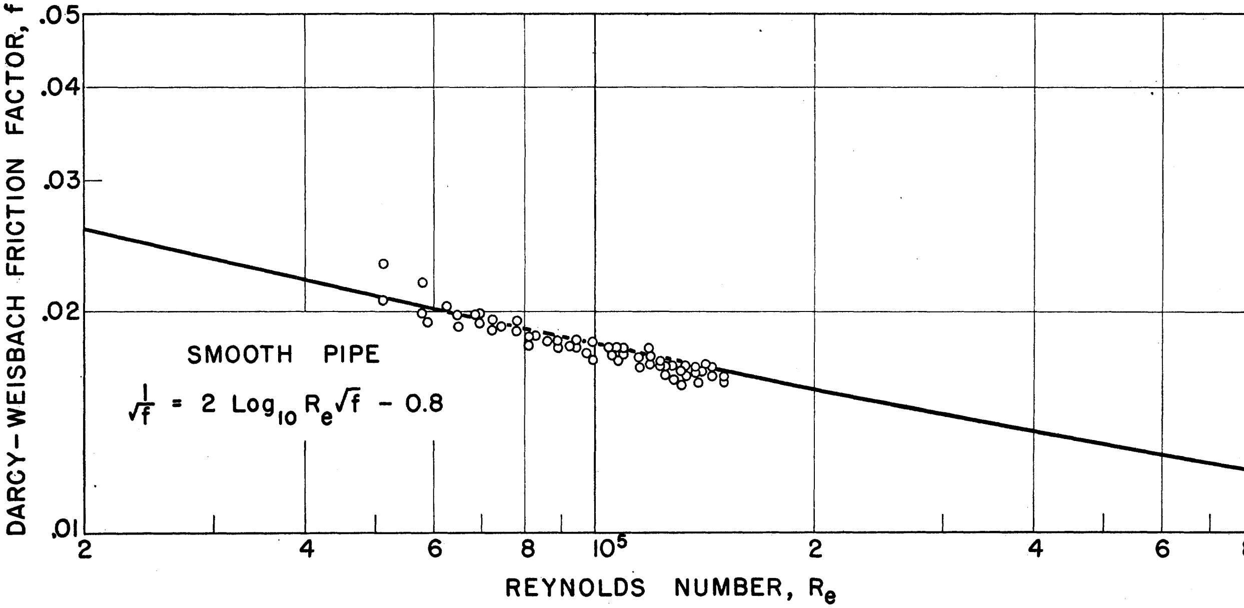

FIGURE 1.—Friction factor versus Reynolds number, for nearly new 3-inch-diameter aluminum tubing. (Experiment by Gray, Levine, and Bogema (*2*).)

Research by H. M. Olson

As reported in his thesis, Olson [2] tested 40-foot lengths of ALCOA 63S–T6 tubing. Four pipes were tested, including

- 3-inch diameter, new
- 4-inch diameter, new
- 4-inch diameter, used
- 5-inch diameter, new

Four couplers were also tested, but the new-pipe tests will be discussed first. The values of the friction factor (*f*) were plotted against Reynolds number (fig. 2). A comparison of these points with the plot of the Karman-Prandtl equation shows the three new pipes to be hydraulically smooth. The scattering of the small Reynolds numbers may have been due to increase in experimental error.

The data from the tests on the used 4-inch pipe are plotted (fig. 3). Comparison with the two Colebrook-White curves shows this particular piece of pipe to have a relative roughness of 0.0003. In his conclusions (p. 20 of the thesis), Olson states: "It is the author's opinion that the roughness of the used tubing is not due entirely to the formation of aluminum oxide tubercules and pitting of the pipe surface but also to many dents in the pipe wall which occur in field handling."

In response to an inquiry, Olson wrote: "I do definitely recall that there were some aluminum oxide tubercules, but they were certainly not of the magnitude one would find on the inside of an untreated steel tubing which had rusted. There were also a number of small dents in the tubing resulting no doubt from field handling." [3]

The experiments on coupler loss will be discussed in the section of this report devoted to that problem.

Research by L. S. Willardson

Tests conducted by Willardson [4] on couplers involved measuring hydraulic grade lines in the two 20-foot lengths of ALCOA 63S–T6 tubing which were joined by the coupler being tested. The description of the pipe states (p. 17 of thesis): "The pipes used for these tests had had little use and should be classified as new pipe." From these observations on both 3- and 4-inch-diameter tubes, he concluded that head loss in the tubing could be expressed by the equation

$$H_f = 0.000246 \frac{L}{D^{1.40}} V^{1.80}.$$

This equation is the result of nearly 300 tests made on 4 different lengths of 4-inch-diameter tubing and 2 lengths of 3-inch-diameter tubing. Rather than use this equation, however, the Darcy-Weisbach friction factors and the corresponding Reynolds numbers were calculated for each of the tests from the basic data. Plottings of the results are illustrated (figs. 4–7).

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[2] OLSON, H. M. THE DETERMINATION OF THE FRICTION FACTOR FOR NEW AND USED ALUMINUM TUBING AND HEAD LOSS IN SPRINKLER-PIPE COUPLERS. 1950. [Unpublished master's thesis. Copy on file in Library, Utah State Univ., Logan.]

[3] In correspondence (Dec. 26, 1955).

[4] WILLARDSON, L. S. ENERGY LOSSES IN ALUMINUM IRRIGATION PIPES DUE TO DEFLECTIONS IN THE COUPLERS. 1955. [Unpublished master's thesis. Copy on file in Library, Utah State Univ., Logan.]

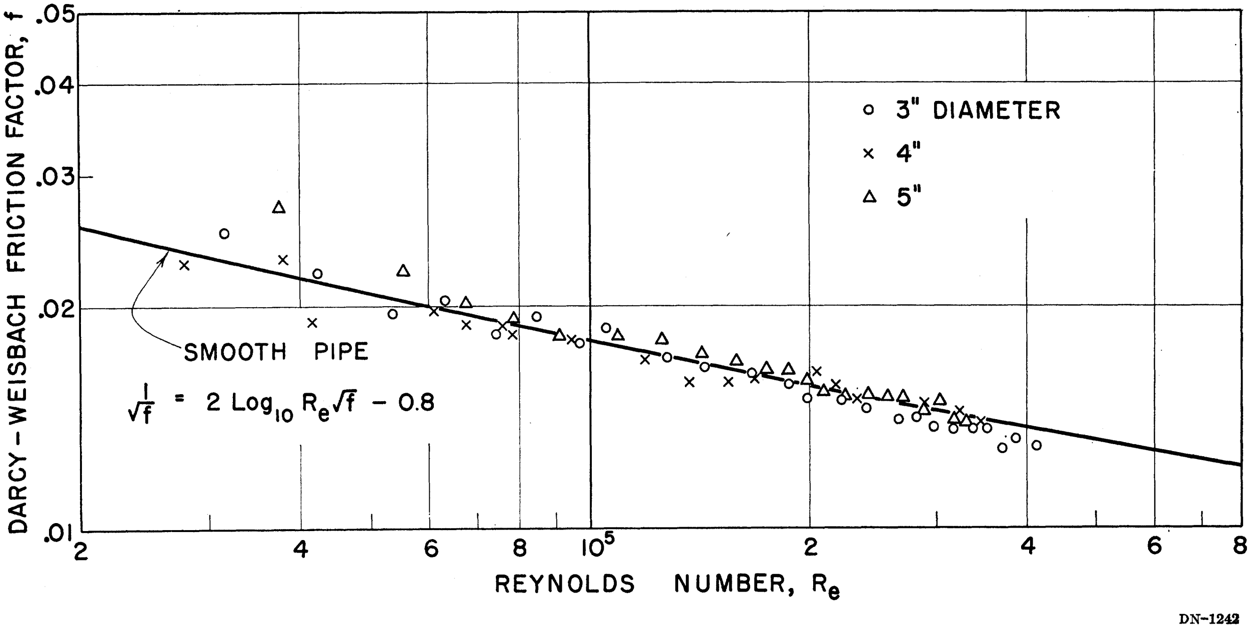

FIGURE 2.—Friction factor versus Reynolds number for new aluminum tubing. (Experiment by H. M. Olson (see footnote 2).)

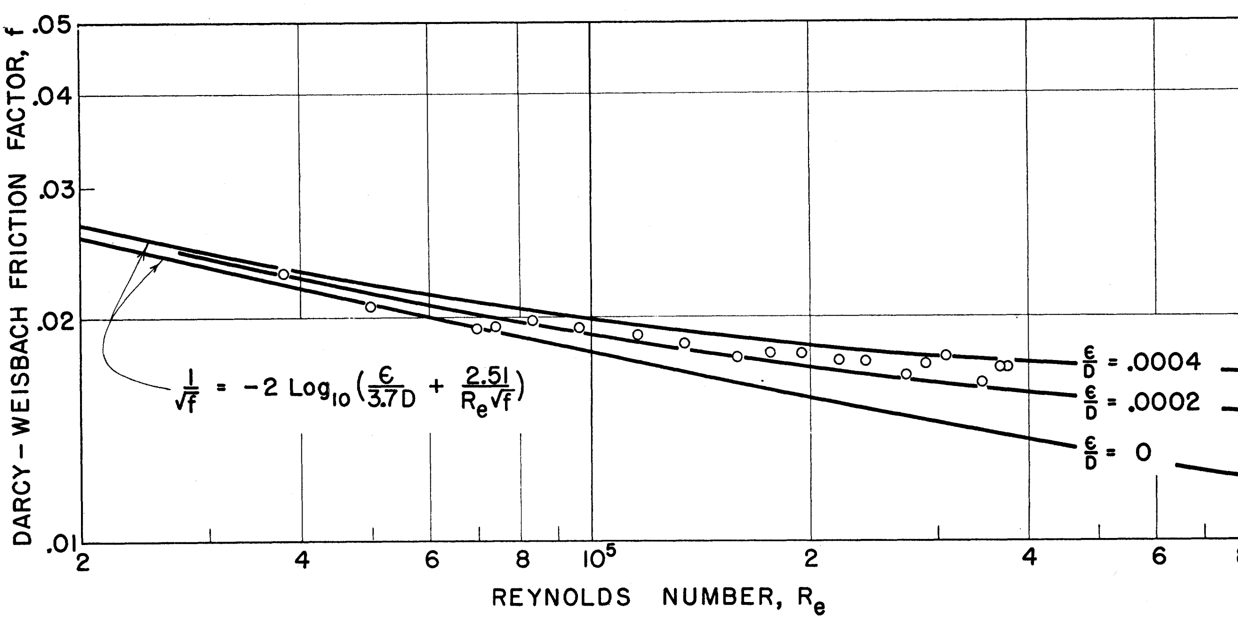

FIGURE 3.—Friction factor versus Reynolds number for dented and corroded 4-inch aluminum tubing. (Experiment by H. M. Olson (see footnote 2).)

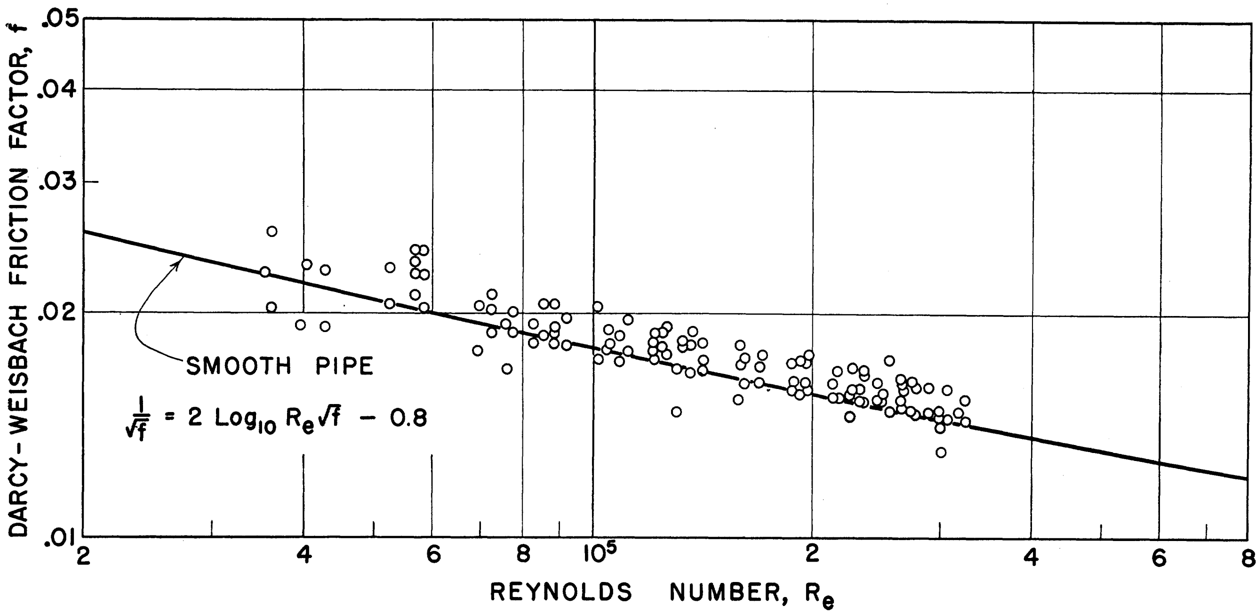

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FIGURE 4.—Friction factor versus Reynolds number for nearly new 4-inch-diameter aluminum tubing, lengths 1 and 2. (Experiment by L. S. Willardson (see footnote 4).)

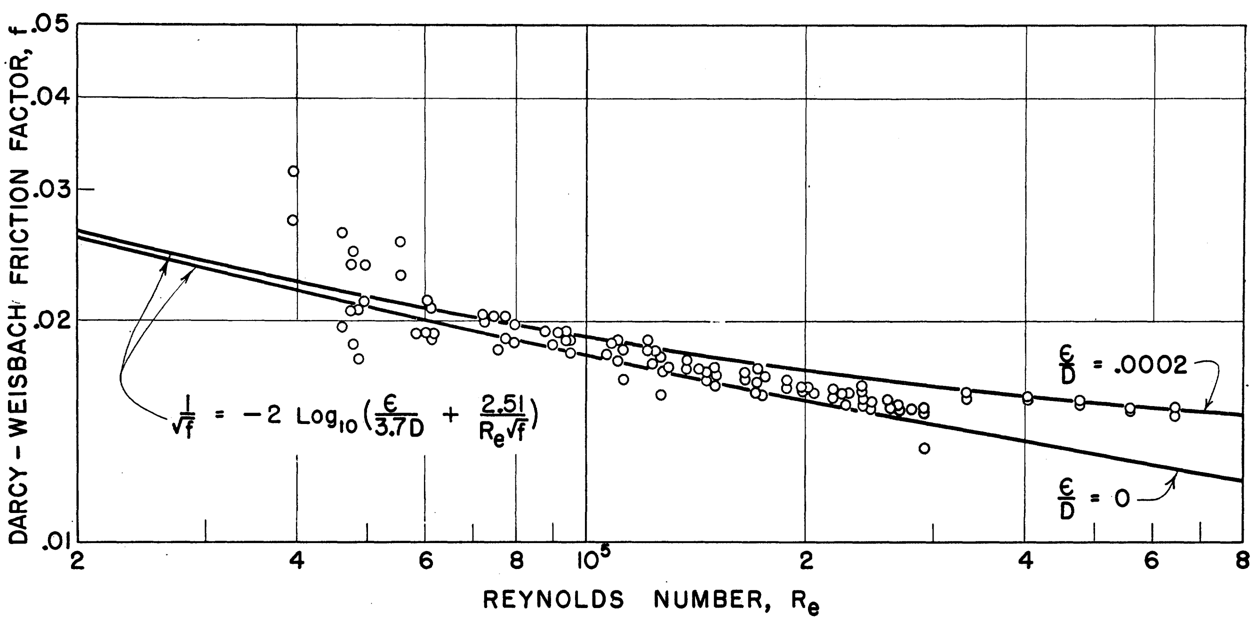

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FIGURE 5.—Friction factor versus Reynolds number for nearly new 4-inch-diameter aluminum tubing. Flow in opposite direction from that in experiment portrayed on figure 4. (Experiment by L. S. Willardson (see footnote 4).)

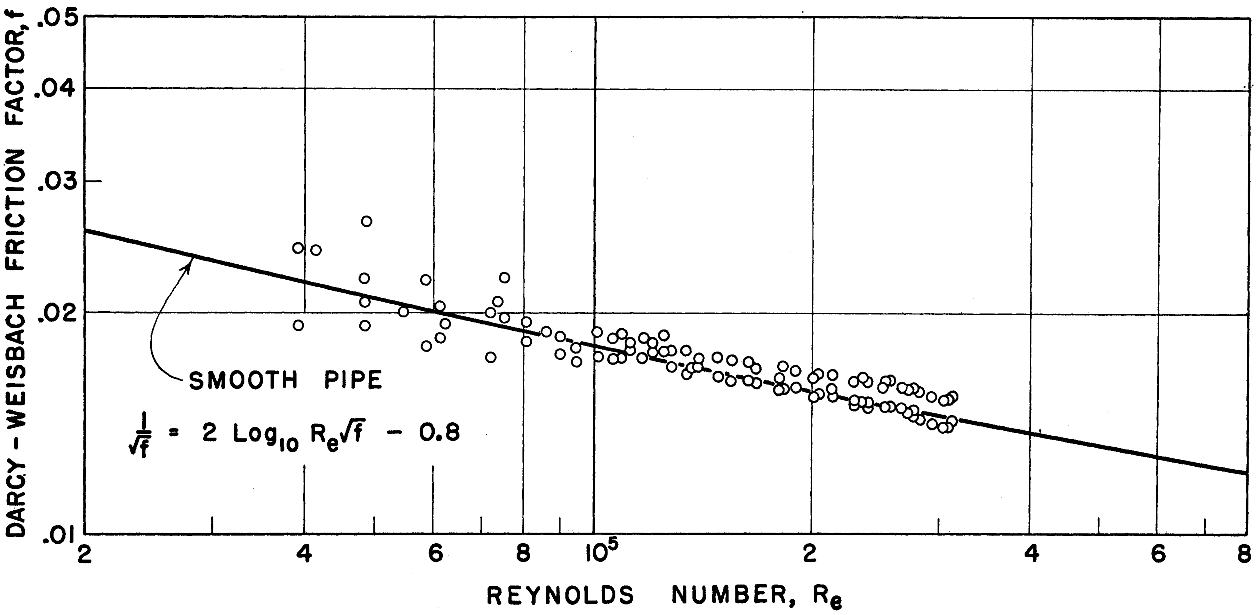

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FIGURE 6.—Friction factor versus Reynolds number for nearly new 4-inch-diameter aluminum tubing, lengths 3 and 4. (Experiment by L. S. Willardson (see footnote 4).)

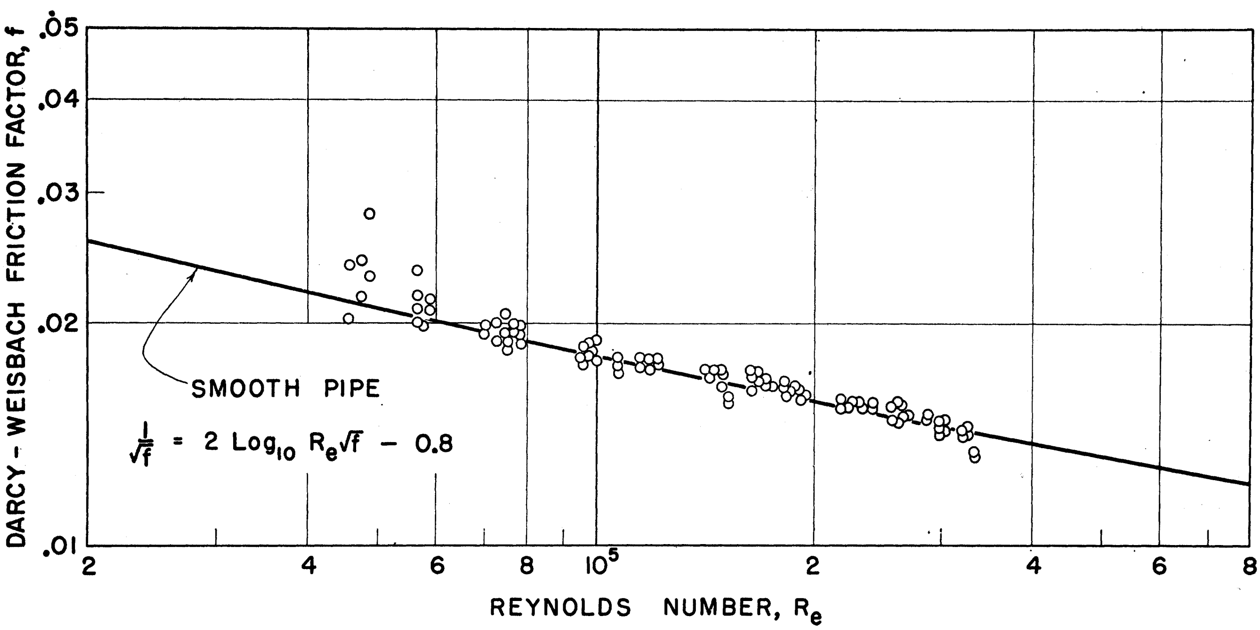

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FIGURE 7.—Friction factor versus Reynolds number for nearly new 3-inch-diameter aluminum tubing. (Experiment by L. S. Willardson (see footnote 4).)

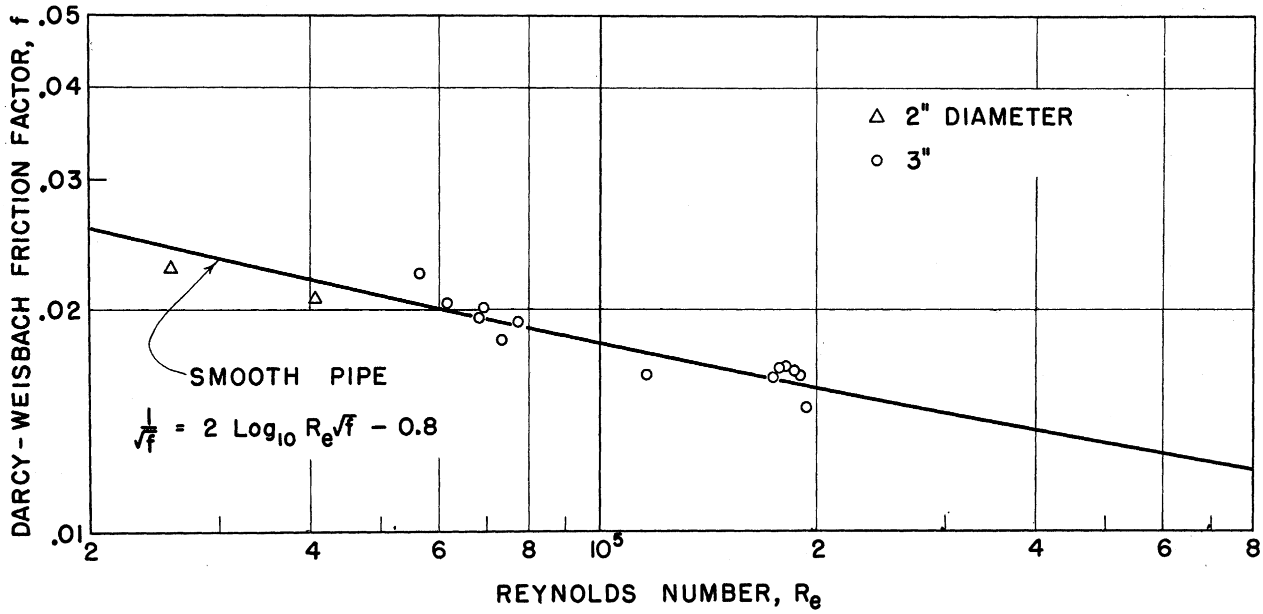

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FIGURE 8.—Friction factor versus Reynolds number for aluminum tubing. (Experiment by A. Benami (see footnote 5) and R. A. Aldrich (see footnote 6).)

A study of these diagrams shows that the pipes appear to be nearly smooth. However, as the majority of the points seem to lie a little above the smooth-pipe curve, it should not be concluded that the pipe is smooth. This is borne out by results shown graphically (fig. 5).

In testing these two pipes, Willardson reached higher values of Reynolds numbers than were reached in all the other experiments. Almost at the very point where the upper values of Reynolds numbers previously reached were exceeded, the resistance function left the smooth-pipe curve and started to follow the Colebrook-White curve for a pipe of relative roughness, $\frac{\epsilon}{D}$ equal to 0.0002. These few tests by Willardson, extending into the higher Reynolds numbers, are very important. They show the pipe to be hydraulically rough. The absolute roughness height is small, however, having an equivalent sand-grain roughness height of only 0.0008 inch.

Data on coupler loss will be discussed in a subsequent section.

Research by A. Benami

In his study of couplers, Benami[5] measured the hydraulic grade lines in the pipe above and below the coupler. These data allowed the calculation of the friction coefficient for the pipes. The pipes were 3-inch aluminum. The Darcy-Weisbach f values were plotted against Reynolds number (fig. 8). It is evident that the pipes are hydraulically smooth.

Research by R. A. Aldrich

Aldrich[6] used a 2-inch aluminum pipe in his experiment. As length over which pressures were measured was rather short, only two of his tests were used to calculate friction factor. The two points are plotted (see fig. 8) along with Benami's data. Here also the points fall near the smooth-pipe curve.

[5] BENAMI, A. EVALUATION OF LOSSES IN BRANCHING-FLOW SPRINKLER COUPLERS. 1954. [Unpublished master's thesis. Copy on file in Library, State Col. of Washington, Pullman.]

[6] ALDRICH, R. A. FLOW OF WATER IN PIPES WITH MULTIPLE OUTLETS. 1952. [Unpublished master's thesis. Copy on file in Library, State Col. of Washington, Pullman.]

EXPERIMENTS ON COUPLER LOSS

The experiments on couplers fall into two classes—those without and those with flow through the sprinkler outlet. The first class yield data useful for main-line pipe design; the other, for the sprinkler laterals.

Research by H. E. Gray, G. Levine, and M. Bogema

Gray and coworkers (*3*) tested 14 different 3-inch quick couplers. Tests were made with the couplers in alined position, then with 12° misalinement (except for 2), and then in offset position where possible. Coefficients K_c were determined for the equation

$$h_c=K_c\frac{V^2}{2g}$$

Where h_c is the head loss in the coupler

$\frac{V^2}{2g}$ is the velocity head.

The results of the coupler tests were effectively portrayed by the use of detailed sketches showing the coupler construction together with the loss coefficients. The coefficient was found to vary with the design of the coupler. For the couplers in alined position, K_c varied from 0.15 to 0.70. In the misalined position, the K_c values increased, the range being 0.16 to 0.84.

Research by H. M. Olson

Olson [7] tested three different makes of couplers, of the following types and sizes:

Type of coupler:	*Diameter of pipe, inches*
A	4
B	3, 4, 5
C	4

These tests were made with no flow through the sprinkler outlet.

Olson observed that the coupler coefficient increased with the velocity. He also found that one make of coupler had a different coefficient for each pipe size. His results are portrayed graphically (fig. 9).

Research by L. S. Willardson

Willardson [8] tested five different couplers of the following types and sizes:

Type of coupler:	*Diameter of pipe, inches*
A, B, C, E	3
A, B, C, D	4

These tests were made with no flow through the riser outlets. The couplers were tested with straight alinement and then with full deflection. Three of the couplers were also tested with flow direction reversed. The results of the experiment are given in the following equation and in table 2.

Coupler coefficient K_c in expression $h_c=K_c\frac{V^2}{2g}$.

TABLE 2.—*Results of coupler tests with no deflection (straight alinement) and with full deflection, in direct and reversed position*

Size of pipe and type of coupler	Deflection angle of coupler	Position of pipeline			
		Direct		Reversed	
		No deflection	Full deflection	No deflection	Full deflection
3-inch:	*Degrees*				
A	11.8	0.108	0.059		
B	5.5	.169	.172		
C	8.1	.062	.105		
E	7.6	.506	.538		
4-inch:					
A	9.7	.110	.064	0.064	0.017
B	5.4	.098	.106	.064	.089
C	8.4	.016		.027	.029
D	13.8	.318	.518		

Research by D. A. Buhr

Buhr [9] tested 5 different couplers. His tests were conducted with flow through the riser. He evaluated the pressure rise across the coupler due to flow out of the outlet. He also measured energy loss between the sprinkler line and riser outlet.

Five couplers of the following types and sizes were tested:

Type of coupler:	*Diameter of pipe, inches*
A	4
B	3, 4, 5
C	4

Analysis has predicted and experiment has verified the rise of pressure that takes place in a pipe when flow passes from upstream of a side outlet to a downstream position. Buhr used the following equation, developed by J. S. McKnown and reported by Soucek and Zelnick (*5*), to predict this pressure rise:

[7] See footnote 2.
[8] See footnote 4.

[9] BUHR, D. A. A STUDY OF HYDRAULIC LOSSES IN SPRINKLER IRRIGATION COUPLERS. 1950. [Unpublished master's thesis. Copy on file in Library, Utah State Univ., Logan.]

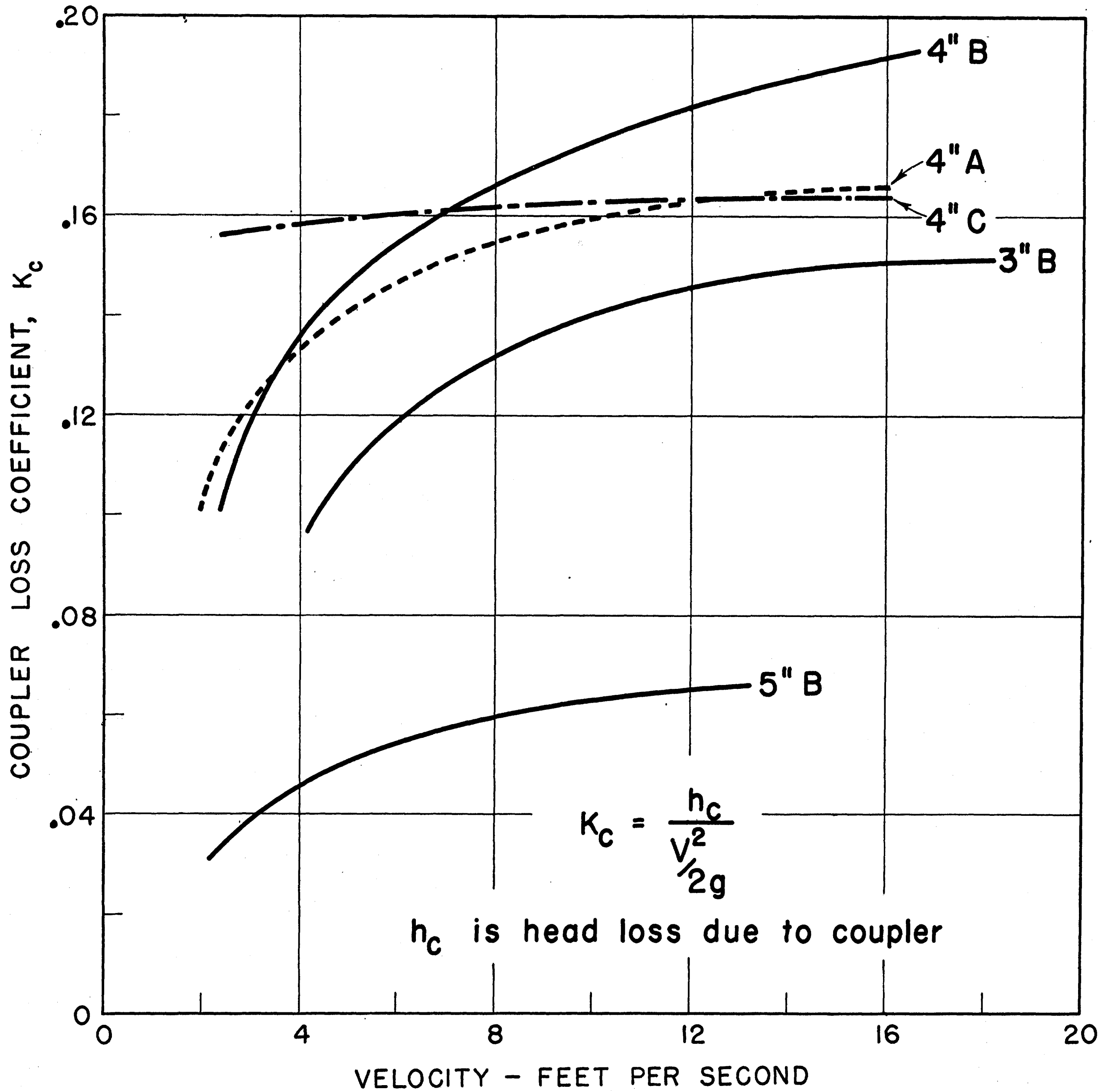

FIGURE 9.—Coupler loss coefficients versus velocity for no discharge through riser outlet. (Experiment by H. M. Olson (see footnote 2).)

$$\frac{\Delta z}{\frac{V_1^2}{2g}}=2R-R^2-\frac{h_f}{\frac{V_1^2}{2g}}$$

Where Δz is the pressure rise in feet

$\frac{V_1^2}{2g}$ is the velocity head of the approaching stream

R is the ratio of the diverted flow to the approaching stream

h_f is the turbulence loss at the junction.

Barton [10] further modified this expression to read:

$$\frac{\Delta z}{\frac{V_1^2}{2g}}=2R-R^2(1+a)$$

Where a is a function of the boundary conditions and is determined by the form of the coupler.

Buhr's results are portrayed graphically (fig. 10).

Research by A. Benami

Tests by Benami [11] were made with flow through the riser outlet. In his findings he separated turbulence loss from the theoretical pressure rise. In this respect he differed from Buhr, who presented the net pressure gain only. The results are shown graphically for two of the couplers (figs. 11 and 12). The turbulence losses were not large.

[10] BARTON, J. R. A STUDY OF DIVERGING FLOW IN PIPE LINES. 1946. [Unpublished master's thesis. Copy on file in Library, State Univ. of Iowa, Iowa City.]

[11] See footnote 5.

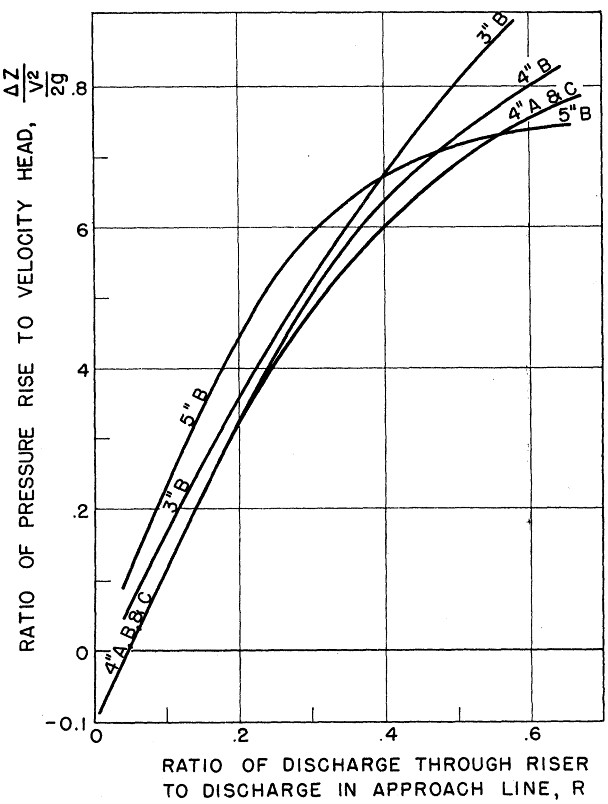

FIGURE 10.—Pressure rise across a sprinkler outlet when flow passes from the upstream to the downstream pipe. (Experiment by D. A. Buhr (see footnote 9).)

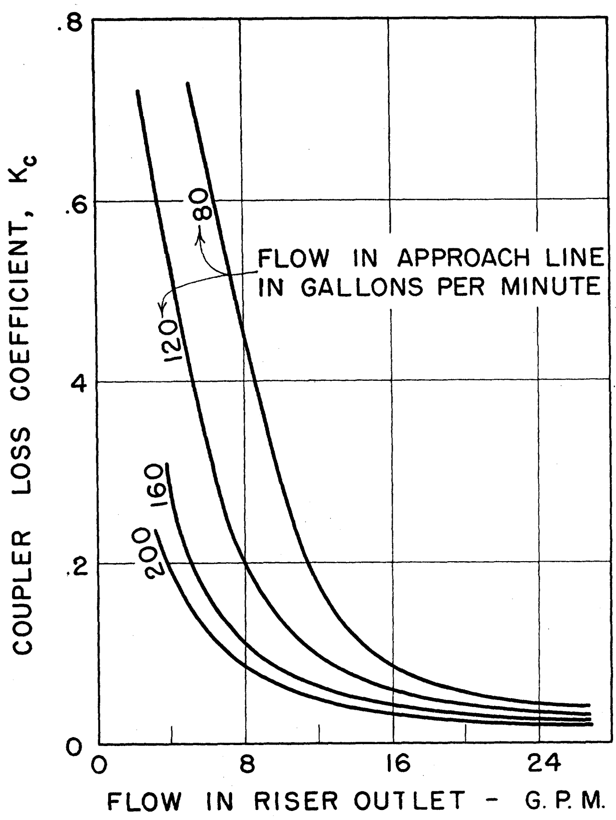

FIGURE 11.—Head-loss coefficients for irrigation-pipe couplers with flow through the riser by coupler No. 1. (Experiment by A. Benami (see footnote 5).)

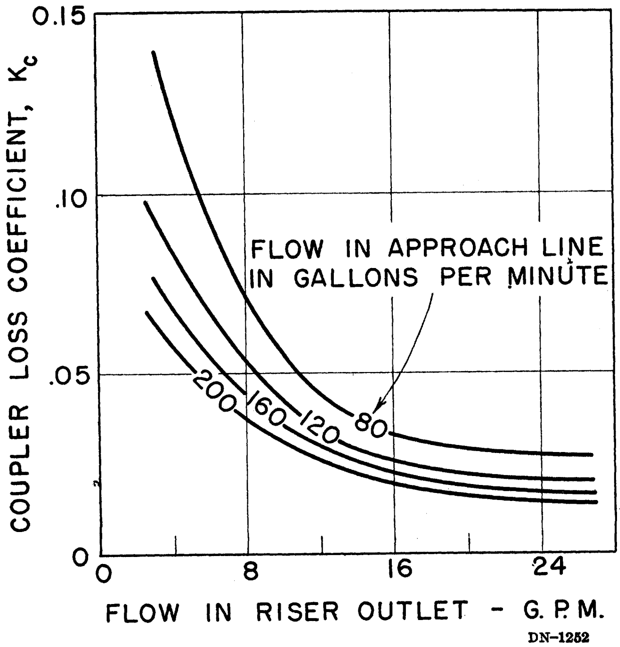

FIGURE 12.—Head-loss coefficients for irrigation-pipe couplers with flow through the riser by coupler No. 2. (Experiment by A. Benami (see footnote 5).)

DISCUSSION OF PIPE FRICTION

At this point in the discussion, an important question should be considered. That is, were the various pipes (except the corroded and dented one) alike? All were described as new or nearly new. In the absence of precise physical measurements of the interior surfaces of the pipes, the only test of similarity lies in the measurements of the friction factors.

The flow tests indicated that differences did exist. Most of the new or nearly new pipes were found to be hydraulically smooth (at least to a Reynolds number of 4×10^5). Yet some of the pipes answering the same description were hydraulically rough. For the purpose of this study, it will be concluded that the new or nearly new pipes tested included both smooth pipe and hydraulically rough pipe.

Adding the corroded and dented pipe gives three categories of pipe to be discussed. These are new pipe, very good used pipe, and poor used pipe. Each kind will be discussed separately in the following sections.

New Pipe

The experiments by Olson, Benami, and Aldrich, discussed previously, showed new aluminum tubing to be hydraulically smooth. This means that the relationship between the Darcy-Weisbach friction factor f and the Reynolds number R_e is best expressed by the Karman-Prandtl equation

$$\frac{1}{\sqrt{f}}=2\log_{10} R_e\sqrt{f}-0.8 \qquad 1$$

This equation can be used to calculate properties of turbulent flow in smooth pipes. However, it has not found general acceptance among hydraulic engineers because most are accustomed to using simpler empirical formulas such as Scobey's, Hazen-Williams', and Manning's. Thus, there is a reluctance to change to a more cumbersome formula even though it might be more exact. As the formulas just mentioned will continue to be used, coefficients for use in these formulas have been calculated for smooth pipe.

The first step in this calculation was to replace the Karman-Prandtl equation by a simple power function. This operation was limited to a narrow range of Reynolds numbers, extending from 10^4 to 10^6. Most flows in portable sprinkler pipe fall in this range, with but few exceeding 4×10^5. Furthermore, a straight line will not deviate far from the curved line of the Karman-Prandtl equation over this range. The equation of the straight line was obtained by calculating the regression equation for the logarithms of selected pairs of f and R_e values. The relation obtained was

$$f=0.2074\ R_e^{-0.2112} \qquad 2$$

By choosing a water temperature (60° F., in this instance), the kinematic viscosity term in Reynolds number reduces to a constant and the equation can be converted to the form

$$H_f=0.295\,\frac{L}{1{,}000}\,\frac{V^{1.79}}{D^{1.21}} \quad \text{---------} \quad 3$$

In this form, the expressed relationship is very similar to Scobey's equation. By the use of this equation, the coefficients were calculated for several of the more common formulas. These will be discussed in turn.

The Scobey Formula

Scobey's formula for metal pipes is

$$H_f=\frac{K_s}{1{,}000}\,\frac{L(V^{1.9})}{D^{1.1}}.$$

By a simultaneous solution of this formula and equation 3, the following expression for K_s was obtained:

$$K_s=0.295\,(VD)^{-0.11} \quad \text{----------} \quad 4$$

If the diameter is expressed in inches, $D=\frac{d}{12}$, then equation 4 becomes

$$K_s=0.388(Vd)^{-0.11} \quad \text{----------} \quad 5$$

Equation 5 has been plotted (fig. 13, the Scobey formula).

From the foregoing equations, it is seen that K_s varies with the velocity as well as the diameter. Consider 15-inch-diameter pipe: K_s will vary from 0.37 to 0.23, depending on the velocity. If the mean value of 0.30 is used for K_s, the error in the estimate of the head loss will be ±23 percent at the extremes of the velocities shown on the diagram.

The Manning Formula

Manning's formula for channels is usually written in the form

$$V=\frac{1.486}{n}\,R^{2/3}S^{1/2}.$$

For pipe flow it can be rearranged to

$$H_f=2.88\,n^2L\,\frac{V^2}{D^{1.33}} \quad \text{-----------} \quad 6$$

A simultaneous solution of equations 3 and 6 yields the following expression for n:

$$n=0.0101\,\frac{D^{0.06}}{V^{0.105}} \quad \text{------------} \quad 7$$

If the diameter is expressed in inches, $D=\frac{d}{12}$, then equation 7 becomes

$$n=0.00870\,\frac{d^{0.06}}{V^{0.105}} \quad \text{-----------} \quad 8$$

Equation 8 is plotted (fig. 13, the Manning formula).

Again, it is found that the coefficient varies with the velocity as well as the diameter. Using the 15-inch-diameter pipe for illustration, the maximum error introduced by a single value choice for Manning's n is ±48 percent. One reason for the large error is that the head loss varies as n^2. (See equation 6.)

The Hazen-Williams Formula

The Hazen-Williams formula is

$$V=C_1R^{0.63}\,S^{0.54}\,0.001^{-0.04}.$$

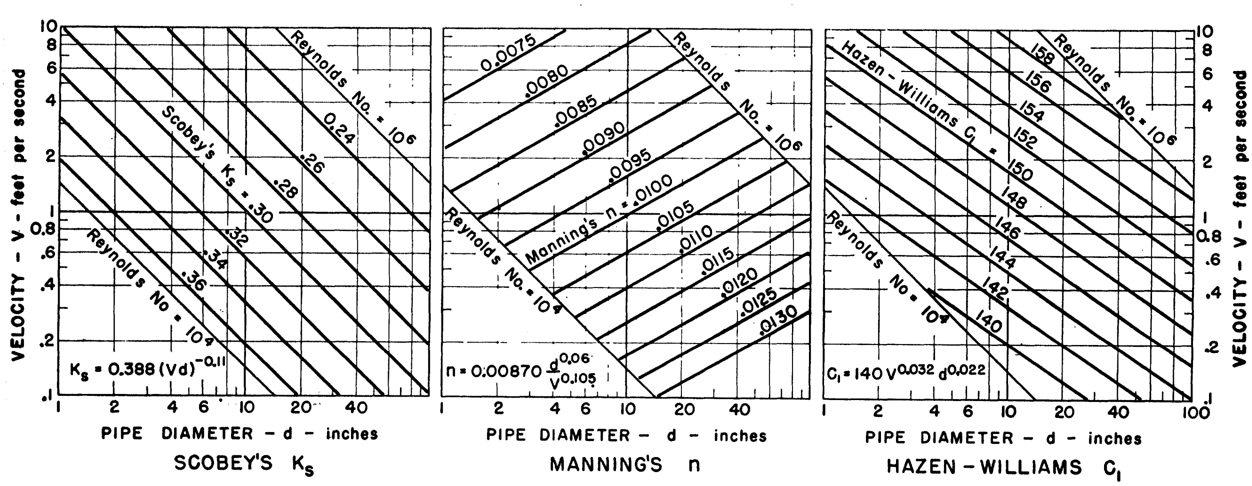

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FIGURE 13.—Coefficient values for the pipe-flow formulas by Scobey, by Manning, and by Hazen-Williams for new smooth aluminum tubing for a water temperature of 60° F.

This can be rewritten as

$$H_f=3.04\,\frac{1}{C_1^{1.851}}\,L\,\frac{V^{1.851}}{D^{1.166}} \quad \text{--------} \quad 9$$

Solving equations 3 and 9 simultaneously yields an expression for C_1 in terms of V and D. It is

$$C_1=148\,V^{0.032}\,D^{0.022} \quad \text{---------} \quad 10$$

For diameter expressed in inches, equation 10 becomes

$$C_1=140\,V^{0.032}d^{0.022} \quad \text{----------} \quad 11$$

Equation 11 is plotted (fig. 13, the Hazen-Williams formula).

Thus it is found that Hazen-Williams C also varies with both velocity and diameter. A single average value of C chosen for the 15-inch-diameter pipe will produce an error in the head-loss estimate of about ± 13 percent. This is over the velocity range of 0.1 to 10.0 feet per second.

Similar derivations could be made to determine coefficients for other formulas for the condition of turbulent flow in smooth pipes. The method is straightforward and proceeds on the basis that the Karman-Prandtl equation correctly expresses the hydraulic characteristics of the pipe.

Very Good Used Pipe

The tests by Willardson[12] showed that some of the nearly new pipe tested appeared to be hydraulically rough. The few tests on two lengths at higher Reynolds number values indicated strongly that these pipes were rough with a relative roughness, $\frac{\epsilon}{D}$, of 0.0002. To study further the departure of friction-factor values from the smooth-pipe curve, all the data for all tests reported here were plotted on one diagram (fig. 14). The Colebrook-White transition curve shown appears to be a fairly good fit for the maximum friction-factor values (excluding the scattered values in the low Reynolds-number range). It is assumed, therefore, that very good used pipe is hydraulically rough with $\frac{\epsilon}{D}$ equal to 0.0002. This assumption permits an analysis similar to that made for smooth pipe.

The first step in this analysis is to fit a straight line to the Colebrook-White curve for $\frac{\epsilon}{D}=0.0002$. In this instance, the fit is limited to a Reynolds number range of 3×10^4 to 7×10^5, which is the range of the data. The equation of the straight line is

$$f=0.107\,R_e^{-0.148}.$$

If a water temperature is chosen, this expression can be rewritten into the head-loss form. For water at 60° F., it becomes

$$H_f=0.000309\,L\,\frac{V^{1.852}}{D^{1.148}} \quad \text{--------} \quad 12$$

Solving equation 12 simultaneously with Scobey's yields an expression for Scobey's K_s for

[12] See footnote 4.

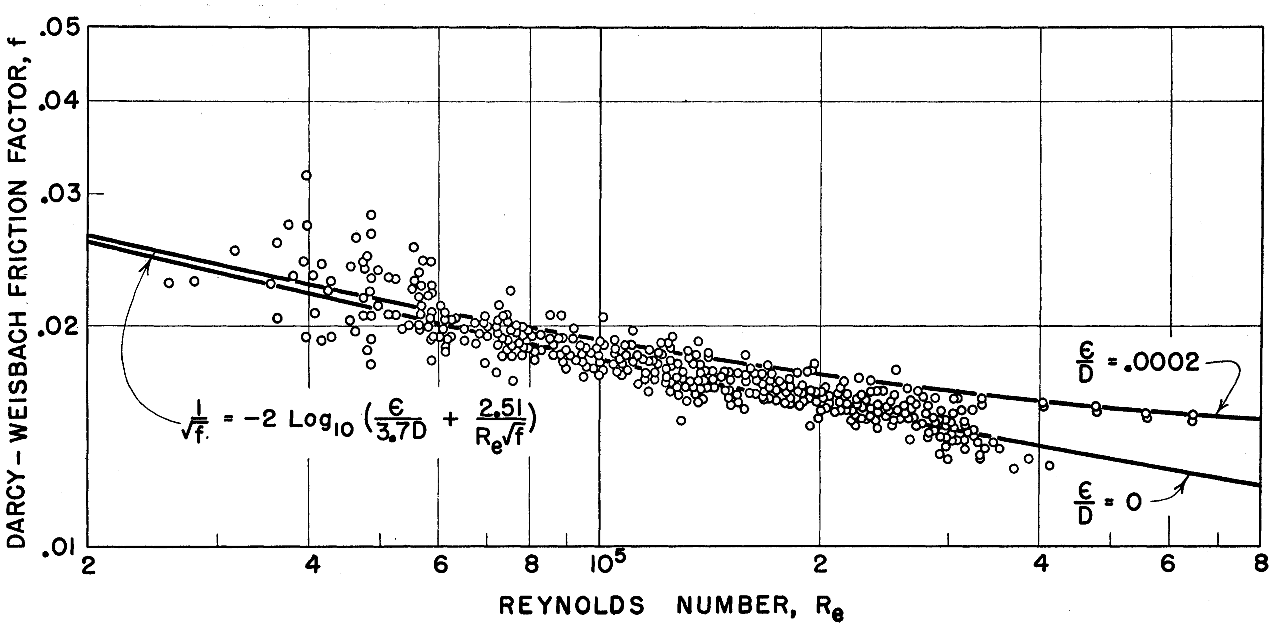

Figure 14.—A composite plotting of friction factor versus Reynolds number for aluminum tubing. (Experiments by Aldrich, Benami, Gray, Olson, and Willardson.)

turbulent flow in hydraulically rough pipes $\left(\frac{\epsilon}{D}=0.0002\right)$. It is

$$0.309\ (VD)^{-0.05} \qquad 13$$

For diameter expressed in inches, this equation becomes

$$0.350\ (Vd)^{-0.05} \qquad 14$$

Equation 14 is plotted (fig. 15, the Scobey formula).

A similar solution has been made for Manning's n. The equation is

$$n=0.0104\frac{D^{0.09}}{V^{0.075}} \qquad 15$$

and for diameter in inches

$$n=0.00829\frac{d^{0.09}}{V^{0.075}} \qquad 16$$

Equation 16 is plotted (fig. 15, the Manning formula).

The solution of equation 12 to obtain an expression for Hazen-Williams C for turbulent flow in rough $\left(\frac{\epsilon}{D}=0.0002\right)$ pipe yields an interesting result. The value of the coefficient is independent of velocity and varies only slightly with diameter. The equations are

$$C_1=143.6\ D^{-0.00972} \qquad 17$$

and (for diameter in inches)

$$C_1=147.1\ d^{-0.00972} \qquad 18$$

Equation 18 is plotted (fig. 15, the Hazen-Williams formula).

It is evident that the Hazen-Williams formula very closely defines the hydraulic characteristics of nearly new aluminum tubing in the Reynolds-number range extending from 3×10^4 to 7×10^5. A value for C_1 of 145 would be satisfactory for very good used irrigation pipe.

Poor Used Pipe

The only data located on aluminum pipe that has had considerable use was in the thesis of Olson.[13] These showed that the relationship between f and R_e followed the Colebrook-White transition equation for rough pipe. The $\frac{\epsilon}{D}$ value turned out to be 0.0003 for the particular piece of pipe tested. Since this was a 4-inch-diameter pipe, the absolute roughness value, ϵ, would then be 0.0012 inch. Therefore, the friction factor for any other diameter of pipe having an interior surface similar to the one tested could be estimated by substituting this value of ϵ and the pipe diameter in the Colebrook-White equation.

A study to determine the values of coefficients for other formulas similar to the study on new pipe was not attempted. It could be done for selected values of ϵ, the absolute roughness height, if it were assumed that the Colebrook-White equation applies. The evidence indicates that it does apply. However, in the absence of data relating ϵ to measurable or definable physical characteristics of the interior pipe surface, the effort expended in such a study would not seem justified. When additional data become available, such a study could be made.

If research is undertaken to evaluate f for used pipe, the value of ϵ should be determined and related to an adequate physical description of the pipe. These tests could be made satisfactorily on

[13] See footnote 2.

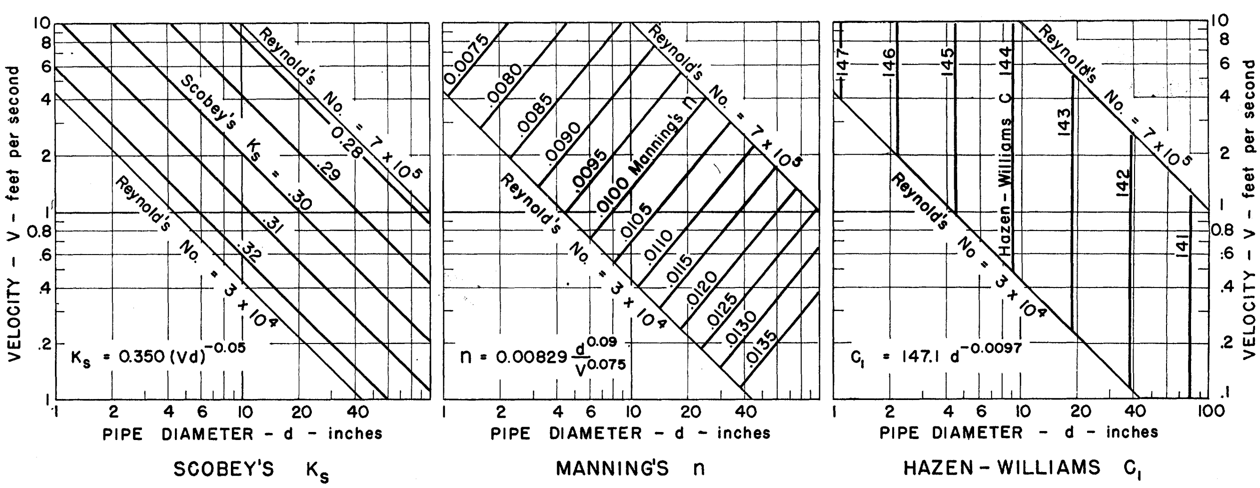

FIGURE 15.—Coefficient values for pipe-flow formulas by Scobey, by Manning, and by Hazen-Williams for very good used aluminum tubing with a relative roughness of 0.0002 and for a water temperature of 60° F.

single 40-foot or even 30-foot lengths of pipe when tests are made with precise measurements under laboratory conditions. In any such experiments, Reynolds number should reach at least 5×10^5 in order to get a good determination of ϵ.

Formula Comparison

Should the need arise to compare coefficients for different formulas, the diagrams (figs. 13 and 15) would be helpful. The only way such a comparison could be made is on the basis of like velocity and like diameter. A study of these diagrams will confirm this.

The question is sometimes asked: Which of the pipe-flow formulas is best? All of the empirical formulas are equally good. The accuracy of the estimate—of head loss, for instance—will depend on the ability to choose the correct coefficient. The study of the three most common formulas has indicated that the Hazen-Williams formula is less sensitive to errors in coefficient selection and may therefore be somewhat superior to the others.

Although the Hazen-Williams formula seems to fit the data a little more accurately, the Scobey formula is used more frequently. Therefore the results of this study are summarized in a few suggested values for Scobey's K_s (table 3).

TABLE 3.—*Suggested values for Scobey's* K_s *for aluminum tubing used in irrigation (without couplers)*

Diameter of pipe	New pipe, smooth	Very good used pipe
Inches		
3	0.30	0.31
4	.29	.31
6	.28	.30
8	.27	.30
10	.26	.29
12	.26	.29

The question is sometimes raised as to what diameter should be used when calculating head loss by use of formulas. Should it be the nominal diameter or the actual inside diameter? The diameter used in the determination of the coefficient should be employed in subsequent calculations which use the coefficient. The studies reported here all used the true inside diameter of the pipe in the coefficient determinations.

DISCUSSION OF COUPLER LOSS

The coupler head-loss coefficient was found to vary considerably with the form of the coupler. Gray's tests showed that the coefficient ranged from 0.15 to 0.70 for 14 different makes, all of the same size. The tests by Olson [14] showed that size of coupler also affected the coefficient. One type of coupler, designated as *B* in this report, at a flow velocity of 5 feet per second (f. p. s.) showed the following loss coefficients:

Diameter of pipe, inches:	Loss coefficient K_c
3	0.11
4	.15
5	.05

Willardson's tests [15] also showed size to be a factor. The coefficients for 2 sizes of 3 different types of couplers are shown (table 4).

TABLE 4.—*Effect of size on the coupler coefficient for 3 different types of couplers*

Type of coupler	Coefficient when diameter of pipe was—	
	4-inch	3-inch
A	0.11	0.11
B	.10	.17
C	.016	.062

[14] See footnote 2.
[15] See footnote 4.

Olson also found that the coefficient varied with the velocity. The 4-inch coupler-loss coefficient ranged from 0.12 at 3 f. p. s. to 0.19 at 16 f. p. s. The results obtained by Gray (*2*) and by Olson cannot be compared, since a common type and size of coupler was not tested by each.

Once a coupler head-loss coefficient has been selected, the next problem is to introduce it into the calculations. One suggested method is to add an amount to the pipe friction factor so that coupler head loss will be included in the friction head-loss estimate. Equivalent values of Scobey's K_s for the coupler loss can be determined by using the following equations:

Head loss for 1 coupler is

$$H_c(1) = K_c \frac{V^2}{2g}.$$

Total head loss for couplers on a line of length L and a spacing S is

$$H_c = \frac{L}{S} K_c \frac{V^2}{2g}.$$

Therefore, the equivalence sought is

$$\frac{L}{S} K_c \frac{V^2}{2g} = K_s \frac{L}{1{,}000} \frac{V^{1.9}}{D^{1.1}}.$$

By using algebra and by assuming a water temperature of 60° F., this is reduced to

$$K_s = K_c \frac{5.00}{S} D R_e^{0.1}.$$

Thus, the equivalent value of Scobey's K_s depends not only on the coupler coefficient but also on the coupler spacing and the pipe diameter.

Values of equivalent K_s have been calculated for a Reynolds number of 10^5 and are presented (table 5).

TABLE 5.—*Equivalent values of Scobey's* K_s *for coupler-loss coefficients* K_c

Spacing (feet)	Diameter of pipe	Equivalent K_s		
		When K_c=0.1	When K_c=0.2	When K_c=0.3
	Inches			
20	3	0. 02	0. 04	0. 06
30	3	. 01	. 03	. 04
40	3	. 01	. 02	. 03
20	6	. 04	. 08	. 12
30	6	. 03	. 05	. 08
40	6	. 02	. 04	. 06
20	12	. 08	. 16	. 24
30	12	. 05	. 11	. 16
40	12	. 04	. 08	. 12

A comparison of these values with Scobey's K_s for pipe friction loss will indicate the relative importance of coupler head loss in the flow. Also, this table shows the special need for hydraulically efficient couplers in the larger diameter pipe lines.

The tests of Gray, Olson, and Willardson were all made *without* flow through the coupler sprinkler outlet. The results therefore are limited to direct application to mains or other lines without side outlets. The experiments by Benami [15a] and Buhr [16] were conducted *with* flow through the side outlet. Buhr presented his results in the form of pressure rises across the outlets. Benami separated the turbulence loss from the pressure rise due to velocity-head change and thus derived the coupler-loss coefficients. Benami's work was especially interesting, because he showed that the coupler-loss coefficient varied with the flow, and with the ratio of the flow diverted at the outlet to the flow of the approaching stream.

The research on couplers to date has been of great help in estimating coupler losses, as it has provided data where none were available heretofore. At the same time, however, such research has indicated that a number of variables affect the coupler-loss coefficient, thus introducing new problems and questions. More data are needed; and simpler methods of applying the data already obtained to design problems in the field are required.

[15a] See footnote 5.
[16] See footnote 9.

DISCUSSION OF HEAD-LOSS TABLES NOW IN USE

The literature of the irrigation industry [17] has been examined. Head-loss tables in these publications have been compared. Comparative values for 1 pipe size—4-inch-diameter—are given in a composite table (table 6).

A study of the following table shows the various sources to be in fair agreement. The comparison, however, raises questions. The first question concerns the discrepancies in the descriptive material provided for column 1 of the table. None of the four sources using the identical table agree exactly on the conditions or kind of coefficient. A check calculation showed that the values in this column were calculated by use of Scobey's formula with K_s=0.32 and an inside diameter of 0.323 feet (this is 4-inch O. D. tubing with a wall thickness of 0.063 inch).

Another question raised is whether a Scobey's K of 0.32 includes pipe coupler loss. If the pipe is hydraulically smooth, coupler loss must be included to bring head loss up to the tabular values. If the pipe is rough (relative roughness 0.0003), a small coupler loss must be added. This is discussed further in the next section.

[17] The mention in this publication of a trade product or its manufacturer does not imply its endorsement by the U. S. Department of Agriculture over similar products or manufacturers not named.

TABLE 6.—*A comparison of values of head loss (in feet per 100 feet of pipe) for portable sprinkler pipe for 4-inch O. D. aluminum tubing*

Rate of flow (gallons per minute)	Loss reported according to source reference [1]				
	(1)	(2)	(3)	(4)	(5)
	Feet per hundred feet	*Feet per hundred feet*	*Feet per hundred feet*	*Feet per hundred feet*	*Feet per hundred feet*
20	0. 04	0. 03	0. 04	0. 04	
30	. 08	. 07	. 08	. 08	
40	. 13	. 11	. 13	. 13	
50	. 20	. 18	. 20	. 19	
60	. 28	. 25	. 3	. 27	
70	. 38	. 34	. 4	. 37	
80	. 49	. 44	. 5	. 48	
90	. 60	. 54	. 6	. 58	
100	. 74	. 66	. 7	. 72	0. 92
120	1. 06	. 95	1. 0	1. 03	
140	1. 41	1. 27	1. 4	1. 37	
160	1. 82	1. 64	1. 8	1. 77	
180	2. 27	2. 04	2. 2	2. 20	
200	2. 78	2. 50	2. 7	2. 70	3. 00
220	3. 31	2. 98	3. 2	3. 21	
240	3. 91	3. 52	3. 8	3. 79	
260	4. 56	4. 10	4. 4	4. 42	
280	5. 26	4. 74	5. 1	5. 10	
300	5. 98	5. 38	5. 8	5. 80	6. 24

See footnote at end of table.

TABLE 6.—*A comparison of values of head loss (in feet per 100 feet of pipe) for portable sprinkler pipe for 4-inch O. D. aluminum tubing*—Con.

Rate of flow (gallons per minute)	Loss reported according to source reference [1]				
	(1)	(2)	(3)	(4)	(5)
	Feet per hundred feet	*Feet per hundred feet*	*Feet per hundred feet*	*Feet per hundred feet*	*Feet per hundred feet*
350	8. 03	7. 23	7. 8	7. 79	8. 32
400	10. 36	9. 35	10. 0	10. 5	10. 40
450	12. 90	--------	12. 5	12. 51	13. 17
500	15. 73	--------	15. 3	15. 26	15. 94
550	19. 12	--------	18. 6	18. 55	22. 64
600	22. 46	--------	21. 8	21. 79	25. 87
650	26. 10	--------	25. 3	25. 32	--------

[1] Sources are as follows (the data in each instance are accompanied by the explanatory material given):

Column 1.—The data in this column are from the following four sources:

Buckner Manufacturing Co., Inc.
Pipe with couplers. Hazen-Williams formula with $C=120$.

National Rain Bird Sales & Engineering Corp.
Pipe with couplers. Wall thickness, 0.063 inch. Scobey formula with $K_s=0.32$ inch.

Irrigation Equipment Co. [*Steelume*]
Pipe with couplers.

Sprinkler Irrigation Association
Pipe without couplers. Scobey formula with $K_s=0.32$ inch.

(Note that the three sprinkler companies state that the table is for pipe with couplers, whereas the Sprinkler Irrigation Association specifies the same values for pipe without couplers.)

Column 2.—*Food Machinery and Chemical Corp., John Bean Division* [*Shur-Rane Manual*]
Pipe with couplers. Wall thickness, 0.050 inch.

Column 3.—*Irrigation Equipment Co., Inc.*
Pipe with couplers. Scobey formula with $K_s=0.32$ inch.

Column 4.—*Miller and Poston Mfg. Co.*
Pipe with A–M coupler. Wall thickness, 0.063 inch. Scobey formula with $K_s=0.32$ inch.

Column 5.—*Olin Mathieson Chemical Corp.*
Table values were given in pounds per square inch. These have been converted to feet for this tabulation. The corporation advises that the figures do not in any way reflect the friction loss of Olin Mathieson pipe with couplers, but are presented purely as a guide to cover pipe with couplers in general.

HEAD-LOSS TABLES CALCULATED FROM AVAILABLE DATA

By using the data from the experiments described in the preceding section, head-loss values were calculated for 4-inch aluminum tubing (table 7). Extreme variations were used in these calculations in order to arrive at an estimate of what could happen. All calculations in this table are based on an inside pipe diameter of 0.323 feet. This is 4-inch tubing with a 0.063-inch wall thickness.

A study of table 7 shows that calculations based on present industry usage (column 7) are conservative when compared with estimated minimum loss (column 5). When compared with a condition approaching a maximum coupler loss, however, values from present tables fall appreciably below actual loss. For a flow rate of 200 gallons per minute (g. p. m.),[18] the low and high conditions of head loss range from 2.54 to 4.53 feet per hundred feet, when compared with industry-usage calculation of 2.78 feet per hundred feet. The high head-loss condition assumes that couplers are located at 20-foot intervals, with a 12° deflection at each joint. Such crooked alinement seems absurd, but it is included to show what is possible. It may account for some of the difficulties experienced in the field. Even with a straight alinement, however, the coupler chosen would give a total head loss of 4.21 feet, which is still considerably higher than the industry standard.

If the minimum coupler loss were added to the rough-pipe loss, the resulting total head loss would be nearly equal to the industry standard. These figures, however, are not included in this comparison.

Inasmuch as no great difference in head loss exists between the smooth pipe and the rough pipe up to flows of 200 g. p. m., it seems that pipe condition may not be the most important factor contributing to errors in head-loss estimates. Instead, it appears that the inaccuracies occur largely in estimates of coupler loss. Further study on the subject of coupler loss could be undertaken profitably.

One determination of friction loss in a used pipe is an inadequate basis for general conclusions. Additional work should be done to determine the pipe friction characteristics of corroded and dented pipe and to correlate pipe condition with friction factor. A study relating degree of pipe deterioration to time and conditions of usage would also be helpful.

[18] This is a velocity of 5.4 feet per second (f. p. s.) in a 4-inch tube, which is an upper limit for good design. Velocities of 10 f. p. s., however, are sometimes used.

TABLE 7.—*Calculated head loss (in feet per 100 feet of pipe) for various pipes, couplers, and combinations for 4-inch O. D. aluminum tubing* [1]

Rate of flow	(1) Smooth pipe, no coupler	(2) Rough pipe, no coupler	(3) Minimum coupler loss	(4) Maximum coupler loss	(5) Smooth pipe plus minimum coupler loss	(6) Rough pipe plus maximum coupler loss	(7) Present industry usage
Gallons per minute	*Feet per hundred feet*	*Feet per hundred feet*	*Feet per hundred feet*	*Feet per hundred feet*	*Feet per hundred feet*	*Feet per hundred feet*	*Feet per hundred feet*
20	0. 04	0. 04	----------	0. 02	0. 04	0. 06	0. 04
30	. 08	. 08	----------	. 04	. 08	. 12	. 08
40	. 13	. 13	0. 01	. 08	. 14	. 21	. 13
50	. 20	. 20	. 01	. 12	. 21	. 32	. 20
60	. 27	. 28	. 02	. 17	. 29	. 45	. 28
70	. 36	. 37	. 02	. 24	. 38	. 61	. 38
80	. 46	. 48	. 03	. 31	. 49	. 79	. 49
90	. 57	. 59	. 04	. 40	. 61	. 99	. 60
100	. 68	. 71	. 04	. 48	. 72	1. 19	. 74
120	. 95	1. 00	. 06	. 69	1. 01	1. 69	1. 06
140	1. 25	1. 33	. 08	. 94	1. 33	2. 27	1. 41
160	1. 59	1. 72	. 11	1. 23	1. 70	2. 95	1. 82
180	1. 96	2. 14	. 14	1. 56	2. 10	3. 70	2. 27
200	2. 37	2. 61	. 17	1. 92	2. 54	4. 53	2. 78
220	2. 82	3. 13	. 21	2. 33	3. 03	5. 46	3. 31
240	3. 29	3. 68	. 25	2. 77	3. 54	6. 45	3. 91
260	3. 79	4. 27	. 29	3. 25	4. 08	7. 52	4. 56
280	4. 31	4. 92	. 34	3. 77	4. 65	8. 69	5. 26
300	4. 88	5. 62	. 39	4. 33	5. 27	9. 95	5. 98
350	6. 47	7. 51	. 53	5. 88	7. 00	13. 39	8. 03
400	8. 26	9. 73	. 69	7. 71	8. 95	17. 44	10. 36
450	10. 21	12. 15	. 87	9. 74	11. 08	21. 89	12. 90
500	12. 36	14. 85	1. 08	12. 05	13. 44	26. 90	15. 73
550	14. 68	17. 68	1. 30	14. 52	15. 98	32. 20	19. 12
600	17. 23	20. 80	1. 54	17. 30	18. 77	38. 10	22. 46
650	19. 93	24. 30	1. 81	20. 32	21. 74	44. 62	26. 10

[1] Column 1 gives the head loss in feet per 100 feet of new or nearly new pipe. The value of friction factor was taken from the smooth-pipe curve. The evidence indicates that the smooth-pipe curve is the appropriate one to use for this condition.

Column 2 gives the head loss in a used pipe similar to the one tested by Olson (p. 5). This pipe was indicated to be corroded and dented.

Column 3 gives the head loss in feet per 100 feet of pipe contributed by the couplers for a probable minimum condition. This is 1 coupler every 40 feet, with no deflection at the joint and with the loss coefficient assumed to be 0.15, the minimum found by Gray (p. 9).

Column 4 gives the coupler loss per 100 feet of pipe for a maximum loss condition. This calls for 1 coupler every 20 feet, with 12° deflection at the joint, and with the loss coefficient assumed to be 0.84, the maximum found by Gray (p. 9).

Column 5 gives the sum of columns 1 and 3. This is the estimated minimum head loss in feet per 100 feet of quick-coupled 4-inch aluminum tubing.

Column 6 gives the sum of columns 2 and 4. This is an estimate of a possible high head loss in feet per 100 feet of quick-coupled 4-inch aluminum tubing. This is not necessarily a maximum.

Column 7 is identical with column 1 of table 6 and is included for comparison.

FLOW IN A LINE WITH MULTIPLE OUTLETS

Another possible source of error in estimating flow in a sprinkler system is the use of approximate solutions for estimating head loss in a line with multiple outlets. The method most used is the one devised by Christiansen and involves the use of an *F* factor. This method has been described in a previous section (p. 2). To test this possibility, a pressure grade line was calculated in an assumed line by using more exact methods. The line chosen was a 4-inch aluminum line with risers located every 30 feet for a total of 20 risers. The coupler losses were assumed to be what Benami's test indicated. The calculation showed the friction loss to be 4.12 feet. This compares with 4.25 feet estimated by the use of Christiansen's *F* factor. This close agreement is not surprising since the sprinkler discharge varied only 2 percent from the downstream to the upstream riser. Under average conditions, therefore, the approximate method is sufficiently precise for determining head loss in a line with multiple outlets. It was also found that the sum of the theoretical pressure rises across the sprinkler outlets equaled the approach velocity head.

CONCLUSIONS AND RECOMMENDATIONS

A study of available information on head loss in quick-coupled aluminum pipe in sprinkler irrigation systems has resulted in the following conclusions.

1. New aluminum tubing is smooth pipe. The relationship between the Darcy-Weisbach friction factor, f, and Reynolds number, R_e, is given by the Karman-Prandtl equation

$$\frac{1}{\sqrt{f}}=2 \log_{10} R_e\sqrt{f}-0.8.$$

Between Reynolds number of 10^4 and 10^6 and for a water temperature of 60° F., this equation can be satisfactorily approximated by the equation

$$H_f=0.000295L\frac{V^{1.79}}{D^{1.21}}.$$

2. Very good used aluminum tubing is hydraulically rough, but of a low order of roughness. The relative value of $\frac{\epsilon}{D}$ in the Colebrook-White equation is 0.0002. The head loss in this kind of tubing between Reynolds numbers $3\text{x}10^4$ and $7\text{x}10^5$ is approximately

$$H_f=0.000309L\frac{V^{1.85}}{D^{1.15}}.$$

3. By using the equations given in items 1 and 2, the values of the coefficients for the commonly used pipe-flow formulas can be calculated. Diagrams giving the values for the Scobey, Manning, and Hazen-Williams formulas are provided in this report.

4. Suggested values of Scobey's K_s for new pipe and very good used pipe are given (see table 3).

5. A 40-foot length of used 4-inch aluminum tubing described as corroded and dented had a relative roughness of 0.0003. This is the ratio $\frac{\epsilon}{D}$ in the Colebrook-White equation.

6. The tests on quick couplers show loss coefficient values ranging from a low of 0.02 to a high of 0.84. The values depend on the make of the coupler, the size of the pipe, the alinement of the pipes joined at the coupler, and the velocity of the flow.

7. Coupler loss in a pipeline can be expressed in terms of an equivalent Scobey K_s. Values are given in table 5.

Recomendations for future testing of irrigation pipe and fittings are as follows:

(1) Tests should be directed first toward evaluating coupler losses and losses in other fittings. The errors in estimating such losses will likely be greater than the errors in estimating pipe friction losses.

(2) Used pipe should be tested. These tests can be made on 30- or 40-foot lengths. Reynolds numbers of at least $5\text{x}10^5$ should be reached, to get a good determination of the roughness characteristic, ϵ. This roughness characteristic should then be carefully correlated with the physical condition of the pipe.

(3) Consideration should be given to making careful measurements of a few irrigation systems in the field.

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